



Research Department Report

SOME PROPERTIES OF ANTIVIBRATION MOUNTS USED IN BUILDING ISOLATION

C.D. Mathers, M.Sc., C.Eng., M.I.E.E., M.I.O.A. and R. Walker, B.Sc.(Eng.)

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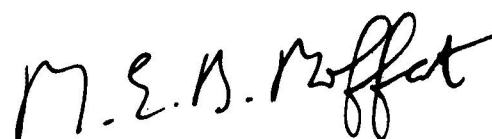
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Summary

The isolation of studios from groundborne vibration in order to achieve a quiet acoustic has been a major concern within the BBC for many years.

About four years ago, a project was undertaken in which the significant factors in building isolation were investigated; one of these factors was found to be the non-ideal behaviour of antivibration mountings (some more than others). This Report describes some of the problems that can be caused by some types of antivibration mounts, particularly those of high intrinsic mass and low internal damping, such as helical steel springs.

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1. INTRODUCTION

Springs or other compliant elements have been used to reduce the forces transmitted by machines or other vibration sources for probably a century. The same technique was applied only much later to isolate buildings or parts of buildings from ground vibration, but is at present rapidly becoming more widespread. The first use of springs (in the form of rubber mat) to isolate a BBC studio dates back about forty years, and seems to have been recognised even then as providing less isolation than expected.

This Report describes two respects in which practical AVMs can deviate from their function as ideal springs or pure compliances: first, modal behaviour, particularly of helical steel springs, which is largely independent of loading or vibration level; second, an experimental demonstration of the effect of friction, which is highly dependent on vibration level.

2. MEASUREMENTS ON ANTIVIBRATION MOUNTS SUPPORTING AN 'IDEAL' MASS

The concept of preventing vibrational energy flow to or from a structure by supporting it on 'antivibration mountings' (a dangerous term if taken too literally) presumably arose from the intuitive picture (quite applicable at low frequencies) of a mass supported upon a compliance. Such a system has a resonant frequency at which it oscillates if disturbed; observing the decay of such oscillations probably gave rise to the idea of viscous damping. The resultant three-element system is shown in Fig. 1(a). By using the electrical analogue^{1,2} of Fig. 1(b), most electrical engineers (like the authors) find the behaviour of the system easier to envisage, and can draw upon wave-filter theory for the analysis of more complex arrangements.

Any practical investigation of a floated structure shows that it deviates in three respects from the three-element model of Fig. 1. These are: modes in the supported structure; modes in the AVM; and acoustic flanking (the transfer in parallel with the AVM of airborne energy to or from the supported structure). The relative magnitudes of these effects for a typical building structure have already been the subject of a detailed investigation^{2,3,4}, which provided both the materials and the opportunity to undertake some 'ideal mass' experiments.

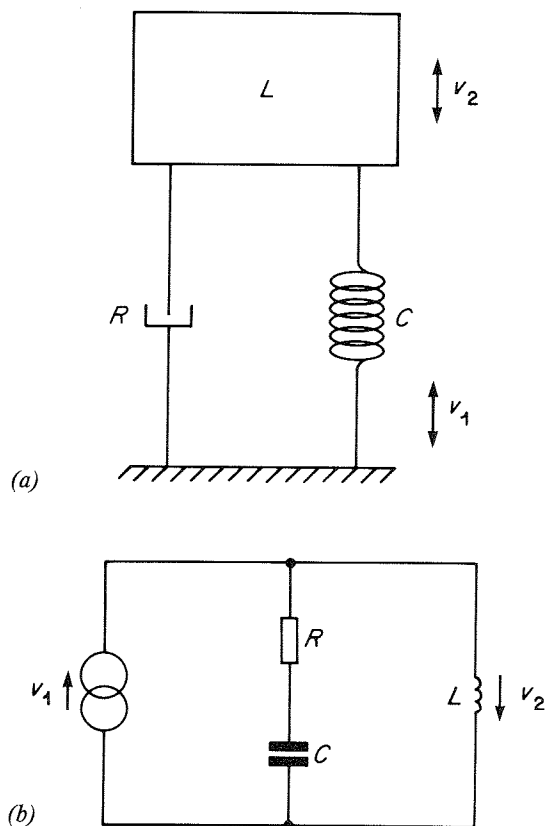


Fig. 1 - Simple model of mass on spring with damping.

(a) Mechanical representation.

(b) Electrical representation.

These experiments were set up using a variety of large AVMs, and a quantity of steel plates which could be bolted together to form a 50 kg mass. The lowest vibration mode of the plates occurred at about 470 Hz, and they therefore behaved as a pure mass at frequencies significantly below this. The mean relative density of a building is of the order of 0.5, whereas that of a piece of steel is about 8; acoustic flanking effects are therefore about 24 dB less for a steel 'ideal mass' than for a building, and may for present purposes be neglected. Thus a very simple setup allowed the behaviour of the AVM itself to be investigated over the frequency range 0 to about 450 Hz without introducing unwanted complications. The only limitation was the maximum load that could safely be placed by one person on the AVM (about 50 kg), which represented only about 1% to 2% of its rated capacity. The purpose of the experiment was to demonstrate the qualitative rather than quantitative behaviour of the various AVM types. Presently, some laboratory experiments and theoretical calculations

will be described which suggest that the behaviour of at least one type of AVM is not qualitatively affected by large changes in loading.

Fig. 2(a) shows the main features of the isolation achieved by the ideal system represented in Fig. 1. The Q factor of the system at resonance depends on the type of AVM, and may vary from 20 or less for a rubber pad to 200 or more for a steel spring. The isolation (expressed in dB) is zero at zero frequency, negative at resonance (indicating an amplification of approximately Q times), then increases at +12 dB per octave from about 1.4 times resonant frequency. At Q times resonance, the slope changes to +6 dB per octave and remains at this value.

The 'ideal mass' experiments cover a frequency range whose upper limit happens always to be less than Q times resonance. Comparison of results is therefore made much easier if a slope of -12 dB per octave is imposed on all plots by dividing the isolation by the square of frequency, so that the useful isolation of an ideal AVM would be represented by a horizontal straight line. The isolation calculated for a three-element system of Q factor 100 at resonance is shown in Fig 2(b) with this correction applied; all experimental results are presented in this form. In all cases (except that of Fig. 8) the 'ideal' isolation, corresponding to the model of Fig. 1, is shown as a dashed horizontal straight line.

3. COMPARISON OF VARIOUS AVM TYPES USING 'IDEAL MASS'

The AVM types measured included rubber in compression, rubber in shear, glass-fibre cubes, and helical steel springs. In Fig. 3 the isolation plots, incorporating the additional slope of -12 dB per octave mentioned above, are compared for the various AVMs using the 50 kg 'ideal mass' described above.

In describing rubber AVMs, the terms 'compression' and 'shear' are enclosed in quotation marks to indicate that because rubber has such a high ratio of bulk to shear moduli, most rubber AVMs, in spite of their appearance, are dependent on the shear rather than the bulk modulus of the material.

The most ideally-behaved AVM tested was a rubber 'compression' type, consisting of a 30 mm thick rubber pad with a thin steel sheet halfway through to increase its stiffness by preventing spreading. The isolation is shown in Fig. 3(a), conforming closely to ideal up to at least 450 Hz.

Fig. 3(b) shows the isolation of a 50 mm rubber-impregnated glass fibre cube, which, although

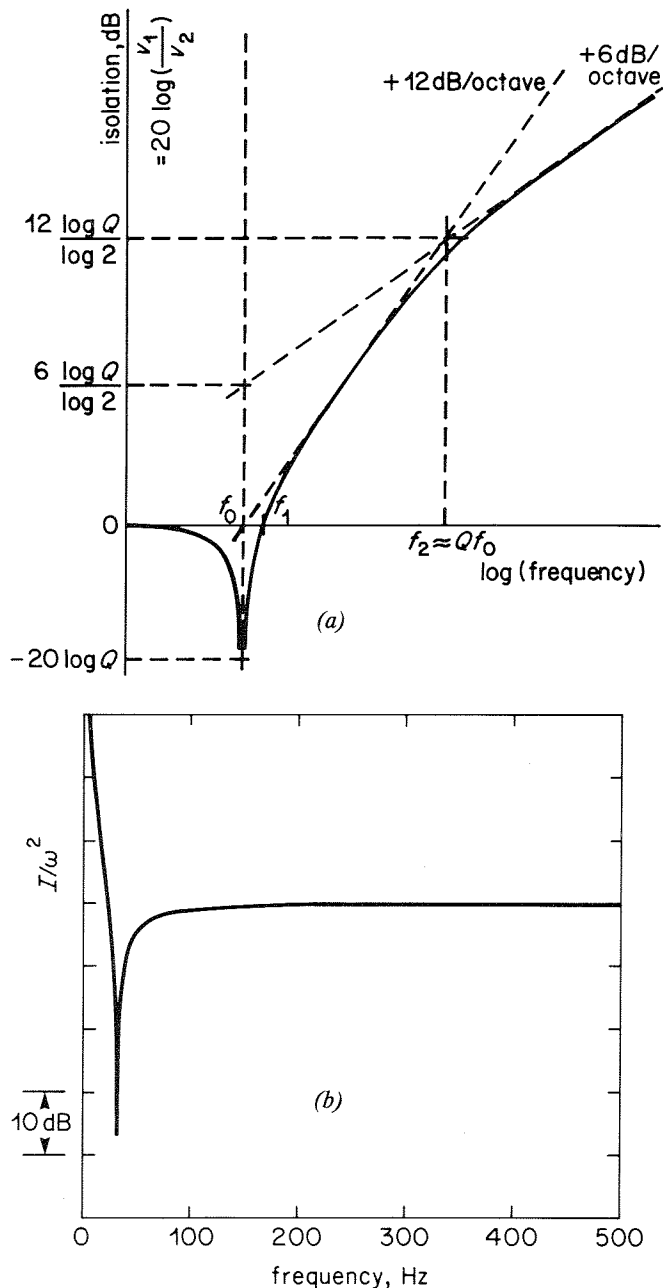


Fig. 2 - Isolation of simple model.

(a) Diagrammatic.

(b) Calculated isolation/frequency² [I/ω^2].

stiff, is extremely light, and is thus very unlikely to exhibit modal behaviour at frequencies below 500 Hz. Nevertheless, in the experimental setup it is somewhat inferior to the much heavier rubber compression mount; this may be due to its much smaller area (50 × 50 mm as opposed to 150 × 100 mm), which would offer less damping to bending or torsional modes of the 'ideal mass'.

It is clear that in a detailed comparative study of AVM behaviour, great care is needed, and that modal analysis of each AVM is probably essential. This impression is confirmed by measuring a rubber 'shear' mount carrying the 'ideal mass'. In this case the

AVM completely dwarfs the stack of steel plates; it has itself a pair of very stiff and heavy steel inserts whose area is much greater than that of such a small load. The measured isolation, shown in Fig. 3(c), is clearly dominated by resonant effects, whose low Q factor suggests that they could well be caused by steel sections embedded in rubber. Sandwiched between a concrete site slab and floor layer, as it would be in any practical application, the shear mount would probably behave in just as exemplary a fashion as the compression type, the end plates being then completely controlled by the surrounding concrete.

Finally, Fig. 3(d) shows the isolation provided by a helical steel spring, complete with 'noise-stop' pad. It is clear that this type of mount deviates most from the 'ideal' behaviour of Fig. 2(b). It is also evident that there are a large number of modes which are not significantly damped by the noise-stop pad, a thin ribbed rubber mat on which the spring's baseplate is placed.

4. FURTHER LABORATORY STUDIES ON SPRINGS

One particular limitation of the experiments described above was the small load that could be placed on the AVMs (typically no more than 2% of maximum). Although it seemed unlikely to the authors that the modal behaviour of springs could be affected other than in numerical detail by even large changes in load, it was decided to set up some laboratory experiments to check the validity of this assumption.

For convenience in measurement, much smaller helical springs were chosen: this meant that they could easily be loaded to a large proportion of their maximum capacity. To avoid the problems of stability inherent in balancing a relatively large load on a single tall AVM, the 'ideal mass', consisting of one or two steel cylinders each weighing 11 kg, was supported on three springs, which it loaded to 23% or 46% of their maximum rating.

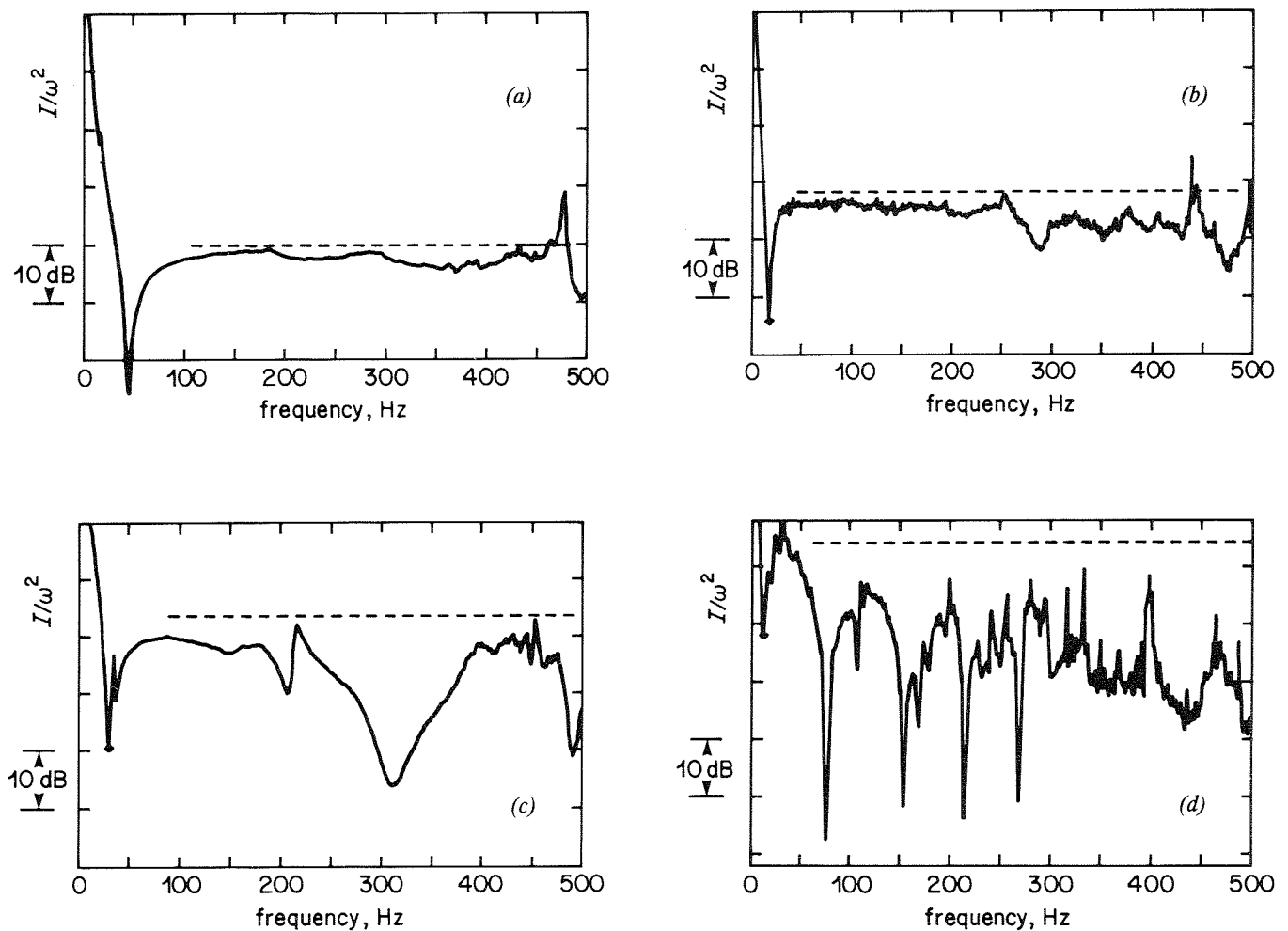
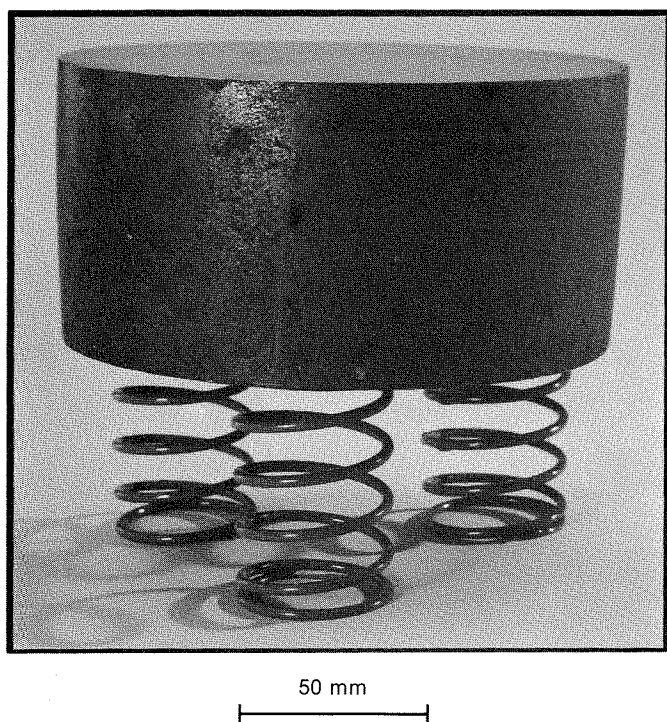


Fig. 3 - Measured I/ω^2 for various AVM types with 'ideal' mass.
 (a) Rubber 'compression' mount. (b) Glass fibre cube. (c) Rubber 'shear' mount.
 (d) Large helical steel spring (with 'noise-stop' pad)

To illustrate the appearance of the laboratory setup, Fig. 4(a) shows a photograph of the 'ideal' mass supported by three springs, and Fig. 4(b) a single spring loaded only by an accelerometer. Under the single spring is a rubber 'noise stop' pad as used in some of the experiments.

Loading one of the springs with the accelerometer alone provided about 3% of its maximum rating, forming a good basis of comparison with the measurements on the large spring. This arrangement also enabled comparison of the modal behaviour of the springs; in this respect, all three were very similar. Fig. 5(a) shows individual isolation plots at 3% loading for the two springs which differed most. The general behaviour closely resembles that of the large spring, shown in Fig. 3(d).



(c). Some modes are significantly damped, whereas others are almost unaffected. The overall isolation is certainly not usefully improved, a finding supported elsewhere⁵.

5. SIMPLE MATHEMATICAL MODEL OF SPRING WITH MODES

To provide a better intuitive understanding of the behaviour of this type of spring, a simple mathematical model was set up to take account of its distributed mass and compliance. The basis of the model is shown in Fig. 6; the spring is divided into a finite number of lumped constituent masses and compliances. As the number of elements N is increased, the calculated isolation of the system

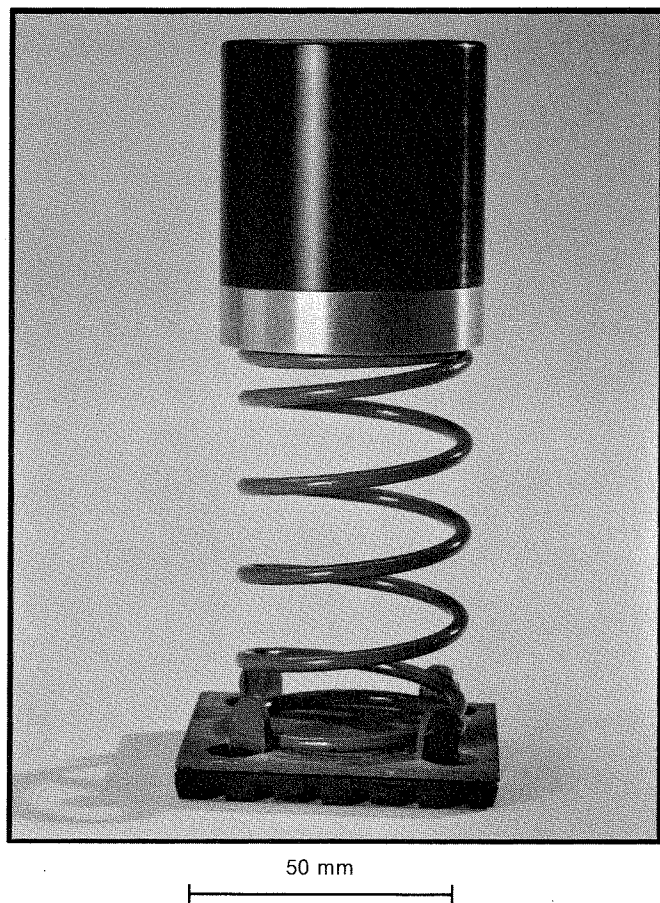


Fig. 4 - Photographs of typical laboratory setups.

Above: 'Ideal' mass supported by three springs.

Right: Single spring with 'noise stop' pad, supporting accelerometer only.

The plots of Fig. 5(b) and (c) show the isolation provided by the three springs together, sharing loads of 23% and 46% of their total capacity. It can be seen that the frequencies of the main series of modes are not greatly affected by changes in load from 3% to 46% of maximum, and that in no case is ideal isolation (indicated by the horizontal dashed lines) closely approached.

The effect of adding 'noise stop' pads can be seen in Fig. 5(d), which is directly comparable with

approaches an asymptotic curve which is independent of N . In practice, a value of $N = 30$ was found to be sufficient.

Fig. 7(a) shows the results of a calculation based on the parameters of the large spring whose measured performance at 2% load appears in Fig. 3(d). Although the modal frequencies are not accurately reproduced, the characteristic arched shape of the curve up to 300 Hz, and the shortfall in isolation compared with 'ideal', are well mimicked. Fig. 7(b)

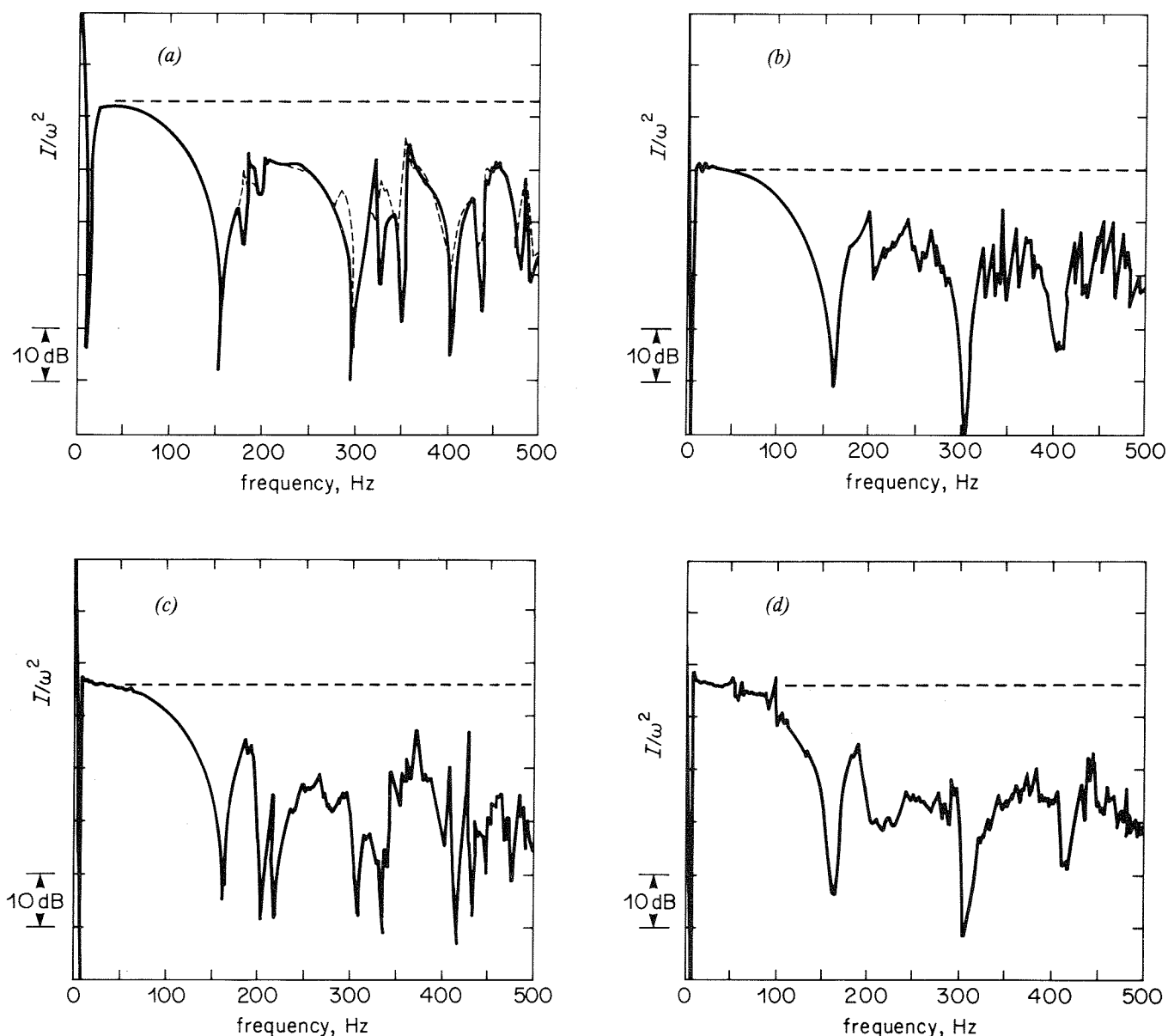


Fig. 5 - Measured I/ω^2 for small helical steel springs with 'ideal' mass.

(a) Two individual springs, 3% load. (b) Three-spring system, 23% total load. (c) Three-spring system, 46% total load. (d) Three-spring system, 46% total load with 'noise-stop' pads.

shows the calculated isolation of the same spring as it would be used in practice, i.e. at 70% rather than 2% of its maximum load. This result agrees with the findings of the laboratory measurements described in Section 4, namely, that modal behaviour is broadly similar at all loadings.

In these simple analyses, no direct appeal has been made to the detailed physical behaviour of springs. In fact, helical springs support three types of mode, corresponding to torsional, transverse, and longitudinal wave propagation; indeed, in Fig. 3(d), at least two modal series are visible. It should however be borne in mind that the purpose of the exercise was

simply to show that the behaviour of a helical steel spring with little damping is indeed dominated by resonance modes.

6. EFFECTS OF FRICTION

There are two essential ways to control vibration. By far the best method is to isolate the source, a fact well recognised by the designers of many new transport systems. The alternative, which is certainly the BBC's only remedy in protecting studios from groundborne vibration such as that from underground trains, is to isolate the sensitive areas.

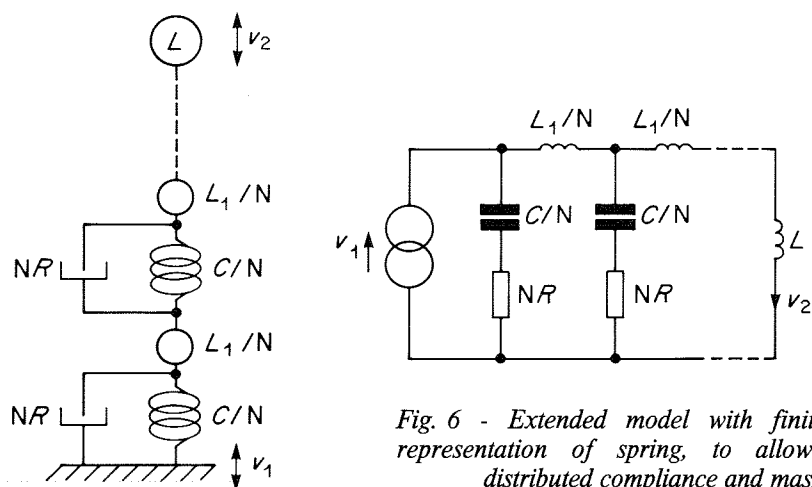


Fig. 6 - Extended model with finite-element representation of spring, to allow for its distributed compliance and mass.

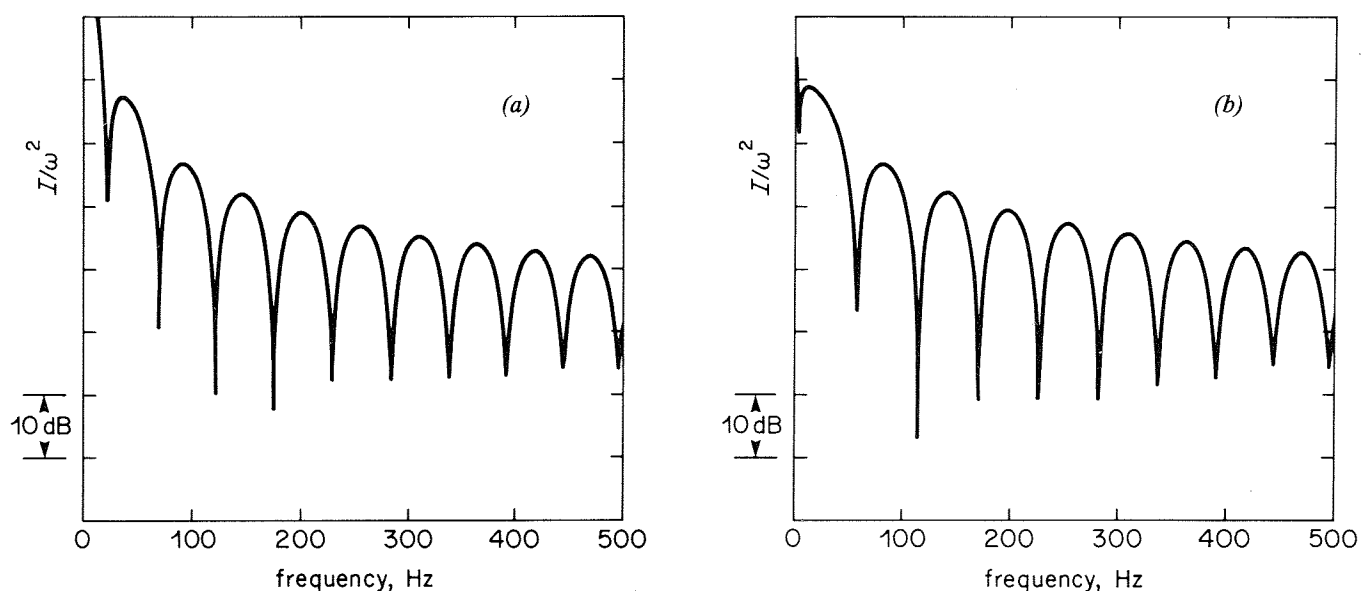


Fig. 7 - I/ω^2 for large spring and 'ideal' mass, calculated from extended model.
(a) 2% load as Fig. 3(d). (b) Expected working load (70%).

There are both similarities and differences between the two approaches. The AVMs are usually identical, as are the theoretical backgrounds governing their behaviour. The typical vibration levels involved, however, may well differ by many orders of magnitude. The forces exerted by a train on uneven rails can be enormous. In contrast, the maximum acceptable vibration level at 100 Hz in the BBC's Maida Vale drama studio (which weighs about 500 tonnes) could be caused by about 50 kgf. When such small forces are important in a large structure, frictional effects can readily play havoc with isolation systems. Mechanical friction is a non-linear phenomenon which prevents the relative movement of two surfaces until some minimum force is exceeded. Thus a degree of residual friction which would be negligible in the suspension system of a train could well

dominate the behaviour of antivibration mountings under a drama studio.

A somewhat dramatic illustration of the effects of friction appears in Fig. 8. The full line is the isolation achieved up to 100 Hz at low vibration levels by the three springs when sharing a single 22 kg mass (each spring being loaded to 23% of its maximum capacity). The dashed line shows the effect of inserting into the coils of each spring a rectangular piece of thin writing paper just shorter than the loaded spring, and of width equal to half of the inner circumference of the spring. The paper is retained in position only by the force needed to bend it into a semicircle (see Fig. 9). With the paper in place, the resonant frequency of the system changes from 4.9 Hz to 6.3 Hz, and the isolation is reduced by about 4 dB. At

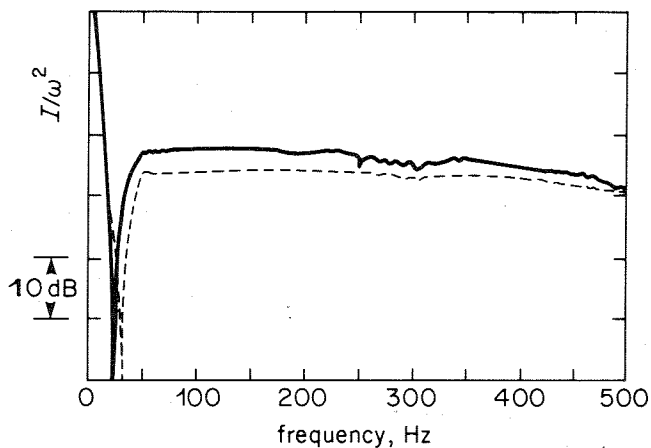


Fig. 8 - Measured I/ω^2 for three-spring system, 23% total load, with and without paper inserted into spring coils.

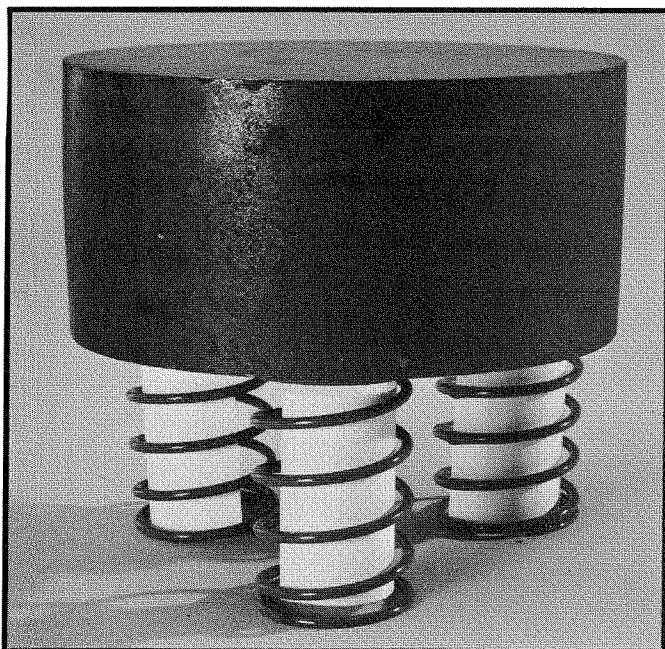


Fig. 9 - Photograph of springs with paper inserted, supporting 'ideal' mass.

higher levels of vibration, the resonant frequency is unaffected by the paper and remains at 4.9 Hz; the isolation also returns to its previous value.

This qualitative demonstration serves to emphasise two points. First, the care that must be taken in the building of structures designed to achieve very low vibration levels, irrespective of the type of AVM used. A brick dropped into a wall cavity, or any pipe or duct which makes frictional contact at its point of entry, could seriously degrade the achieved isolation. Second, any attempts to damp the modes of a helical spring need to be considered with care. For example, the insertion of wire wool may well provide

excellent frictional damping at high vibration levels, but at very low levels may merely appear as an additional compliance, mechanically in parallel with that of the spring.

7. DISCUSSION

Much of this Report has been devoted to the effects of modes in AVMs, of which helical steel springs have been chosen as a particular example. Building structures are also subject to vibrational modes. By the nature of their construction, they are much less predictable than AVMs in their detailed mechanical behaviour, and are very much more difficult to damp. To maximise isolation it is therefore important that not only the AVMs but also the supported structure be as well damped as possible.

It should be borne in mind that above the frequency of its first mode, no AVM or building, however well damped, ever reverts over a finite frequency band to its low-frequency behaviour as a compliance or mass. Thus only at sub-modal frequencies can the ideal performance of Fig. 2 be approached in practice, even in systems with excellent damping. Herein lies the benefit of using intrinsically light AVMs, made of rubber or glass fibre, rather than, for example, heavy helical steel springs.

8. CONCLUSIONS

Comparative assessments have been made of several different types of large antivibration mounting by using them to isolate a compact steel mass whose behaviour closely approximates to ideal (i.e. non-modal) over the frequency range of interest.

As might be expected, the mounts whose behaviour deviated most from that of an ideal compliance were helical steel springs without inter-coil damping. Further laboratory investigations on much smaller springs showed that their modal behaviour was not very dependent on loading, and that the performance of large mounts under a small fraction of their working load provided a reasonable indication of their probable performance in normal use.

The experiments also showed that the placing of a ribbed rubber mat or 'noise-stop' pad under such a spring provided effective damping of only some modes, and that if 'ideal' performance were to be approached, than some more effective form of coil damping would be required. The effects of friction were also demonstrated, indicating that special care was needed in the design and construction of isolation systems for use at low vibration levels.

A simple mathematical model was set up which used a finite-element representation of the spring's distributed compliance and mass, but in which no account was taken of its detailed physical behaviour. This model was sufficient to reproduce the general form of the measured isolation curves, and therefore to confirm that the behaviour of the experimental setup was indeed due to spring modes.

In attempting to isolate buildings or part of buildings from groundborne vibration, it is important to appreciate the factors involved; for example, the air-borne transfer of energy to the 'isolated' structure^{2,3,4}. It is also important to take account of the degree and manner in which the various types of available antivibration mounts deviate from the ideal pure compliance on which isolation calculations are often based. In particular, heavy AVMs with low intrinsic damping, such as helical steel springs, can cause serious shortfalls in the expected isolation if that is based on the very simple mass-compliance-loss model of Fig. 1.

Finally, it must of course be recognised that in practice the choice of AVMs is governed by a number of requirements as well as that of providing adequate

isolation, such as long-term reliability, ease of maintenance, and low cost.

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